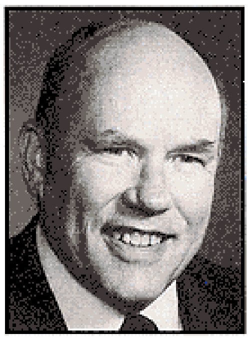
Machinery Messages

Protection against thrust bearing failure in a steam turbine



"...When setting shutdown limits, never try to save the thrust bearing. Save the machine..."

by Charlie Jackson
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Thrust bearings separate the rotating elements of a steam turbine from the non-rotating elements (Figure 1). The intermarriage of these two elements is very costly, with the annulment starting at about $500,000, depending on turbine size. About 80% of the loss will be from lost production.

The best protection is an instrumentation system which is dedicated towards axial deflection detection, with dual-voting eddy current proximity sensors mounted in the turbine's steam inlet section as is shown in Figure 1. Generally, sensors are mounted in the steam end pedestal end plate. These two displacement probes, calibrated to the dc gap voltage, indicate the distance from the sensor tips to the end of the turbine shaft. The end of the turbine shaft must be in close proximity to the "inactive" and "active" thrust bearings, usually 12 inches (300 mm) or less!

Thrust bearings fail quickly (less than 30 seconds) once the supporting hydrodynamic oil film has lost its ability to support the rotor thrust loads. These rotor thrust loads have been recorded to be greater than 500 psi babbitt (white metal) pressure. The metal temperatures from imbedded thermocouples at the 75% arc × 75% radius load zone have been in the 250-285°F. range (sometimes not making it to 250°F. before machinery failure!)

If one wishes to have confirming correlation to data points, then a steam turbine thrust overload will generally have (1) axial movement of the thrust collar/rotor in the damaging range of 15 to 25 mils, plus (2) an increase of active thrust bearing metal temperatures increasing from 170-200°F towards 225-250°F, plus (3) an increase in the first stage pressure by 50 to 100 psig.

NOTE: All these units can vary with size, type, speed, and steam pressure, but are typical for turbines in the 600 psig (700°F) to 1200 psig (900°F) range, from noncondensing to condensing. However, in a noncondensing turbine, contaminants, e.g. salt or silica dew points (500-600°F) may not occur until after the steam exits the exhaust.

The American Petroleum Institute (API) tries to address most of these issues in their API 670 specification. Keep in mind that many USERS make up API and they do not all agree on what comprises a good, better, or best installation. For example, I use the optional reverse mounted proximity probes, but use the dual thrust probe arrangement. I may add an additional, informative probe to record the AC component of the thrust collar run-out, which is typically .5 mil (12 μm) T.I.R. on assembly.

I use the API Calibration of Axial Position (Thrust) Monitor for Steam ►

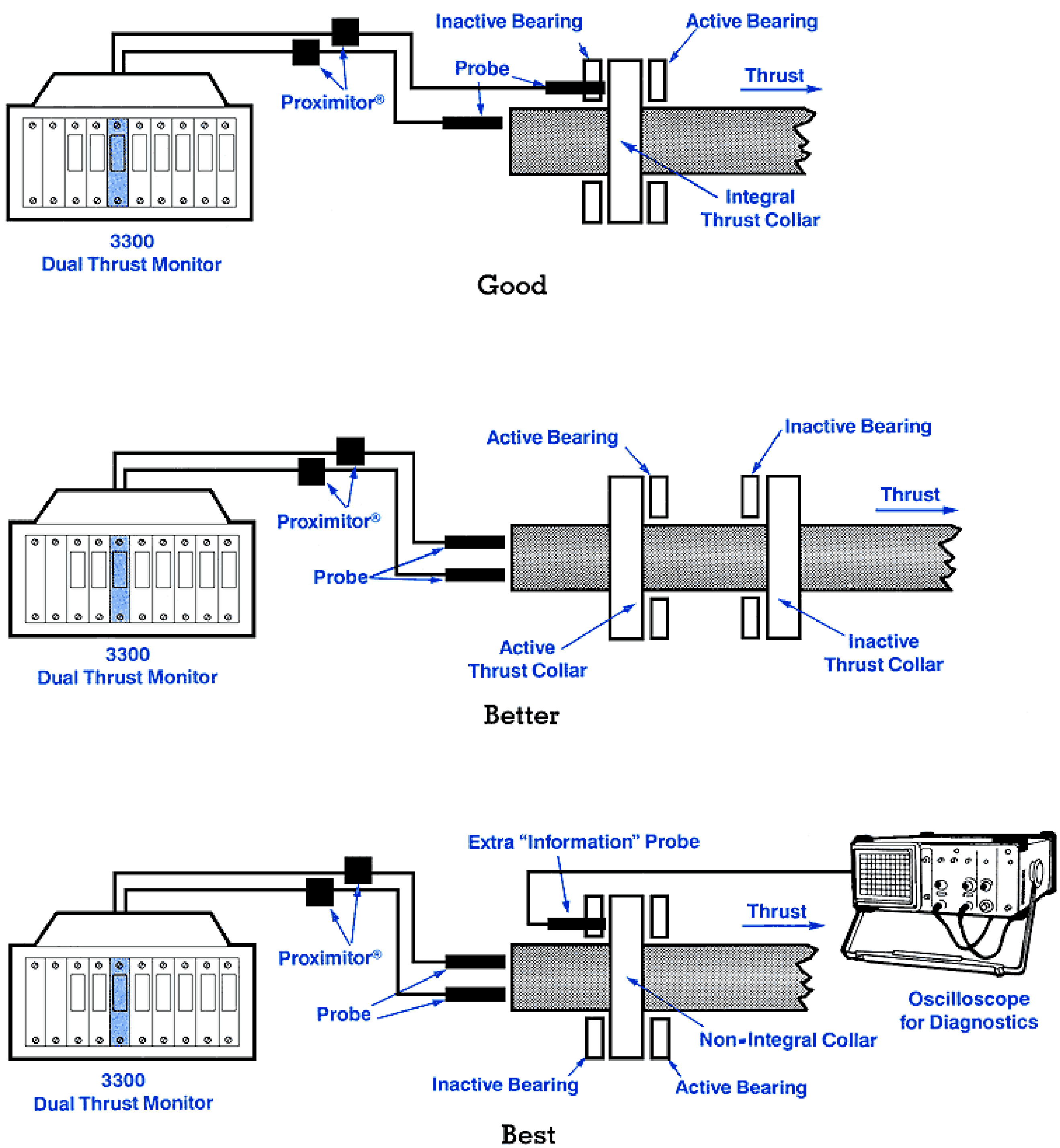

Figure 1
Thrust (Axial) probe arrangements

Turbine chart religiously. I really should use it as it was published in the *Practical Vibration Primer* which I wrote in 1979 but has been in use since 1968. The active thrust position should be the same value within operating units based on the average cold float within the unit, e.g. 16 mils (406 μm). The actual float will vary, and the center of float is never used in setting up the probes or the monitors. With the rotor held against the active thrust bearing (most steam turbines I have seen normally thrust toward the exhaust which is away from the probe), the gap voltage of both probes is set at about –10.8 Vdc corresponding to about 8 mils for a 16 mil (406 μm) float. Now you may say that small turbines have only 12 mils (305 μm) of float and you are correct; so, set +6 mils for normal. Incidently, you should have notified your instrument builder, e.g. Bently Nevada Corporation, to have your monitor read upscale (active or "plus") for a gap opening voltage change (increasing negative dc voltage). If not, you may receive a gap closing versus active versus (+) monitor which is often used for a compressor.

The thrust float zone is defined as the normal allowable movement of the thrust collar within the thrust bearing clearance. However, this float zone will increase when the machine is at full load and operating speed. The change is due to the higher (operating) load on the thrust bearing. Other contributing factors are thermal expansion, springiness of the thrust bearing assembly, thrust pad deflection and squeezing of the oil film. For example, on a 7500 HP steam turbine, the normal design thrust load deflection was 11 mils (279 μm) past "rotor bump check" on startup because the shim pack behind the **active** thrust bearing was 0.3 inches (7.62 mm) thick and was composed of **many very thin shims** which are "spongy" under load.

History: In the early 60's, I retrofitted axial probes to read the thrust collar surface. This was good and provided AC (oscilloscope) information of the run-out indicating any possible shaft bow due to:

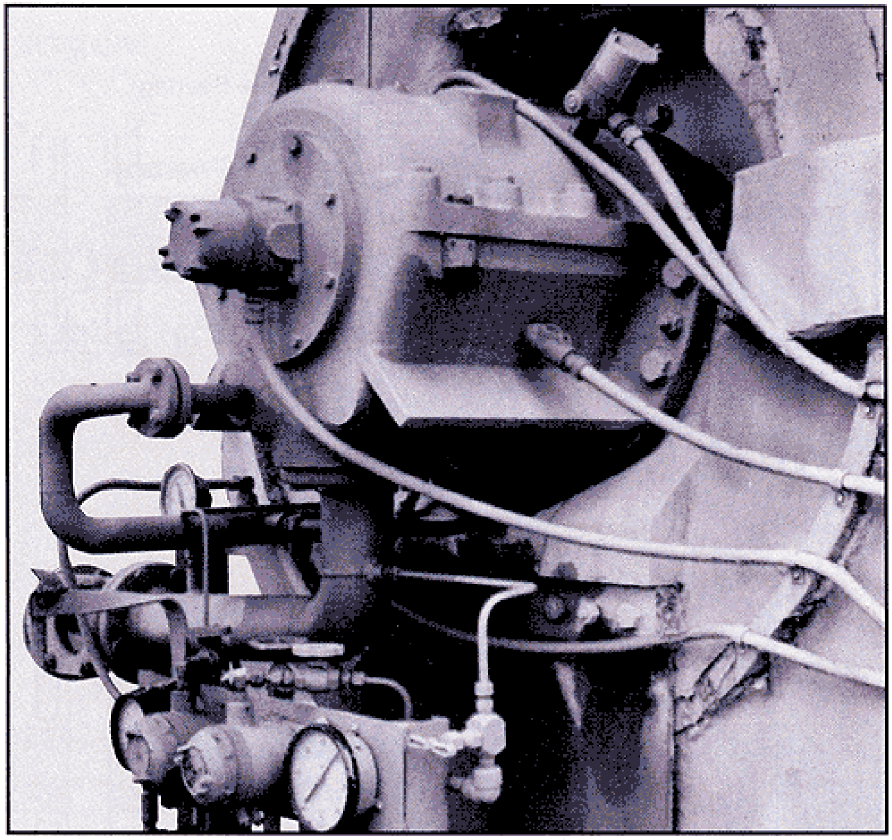

Figure 2
Dual thrust proximity probe machine protection assembly on a centrifugal compressor's discharge end bearing assembly.

1. Temporary bow in the rotor due to long overhauls, long rotor storage, etc. coupled with improper slow roll, heat soak, time on barring gear, etc. (A good soak time is twice the time required to roll out the bow).
2. Permanent bow in rotor due to heavy rub or other damage.
3. Bow in rotor due to introduction of wet steam into the steam inlet for whatever reason (e.g. upset of steam desuperheating station if you really want a "biggie.")
4. Heavy (or light) rub from labyrinths (or carbon rings which is actually more prevalent). Heavy versus light rubs react differently (which is another subject for a different discussion).
5. Failed bearings due to oil failure with rubs and restart.

After hearing of other companies having a typical non-integral thrust collar coming loose and wrecking the turbine without alarm or shutdown, I converted to both axial probes sensing the shaft ends near the thrust collars.

Some users wanted one probe to sense the end of the shaft and the second probe to sense the thrust collar as is shown as the "Optional Installation" in API 670. They feel that the operators will see the one channel in alarm and the other not in alarm, indicating a loose thrust collar and manually shut down the unit. I do not agree with this so I use the "Typical Dual Probe Arrangements" shown in Figure 12 of API 670.

This is much easier to do and I do not have to squeeze a probe between the articulating level links or pads on my thrust bearings. Everyone is entitled to his own opinion.

I feel that:

a. A thrust bearing is damaged before an operator and/or his chief can make a fast enough decision. Less than 30 seconds is less time than you need to mentally analyze the situation.

b. I do not think most operators would know which probe is on the collar and which probe is on the shaft end (except for the operator that might be reading this, who I have to work with on the next job).

c. Most thrust collars are either held by a sleeve or a lock washer/nut. Others are integral collars (large turbines) and some are hydraulically dilated. I know of six incidents where the thrust collar came loose. Two happened to me. The second was fortunate in that, as the lock washer fatigued, it swung out "machete style" and severed the thrust probe coaxial cable shutting the machine down with "same time" Alarm/Danger lights lit.

d. Informative probes are valuable additions for critical condensing turbines.

Example of a proper installation

Figure 2 shows a dual thrust protection assembly on a centrifugal compressor's discharge end bearing assembly. The radial vibration probes can be seen mounted in the standard API 670 XY configuration. The compressor incorporates a balancing piston chamber. This chamber enables the compressor to partially counterbalance the cumulative thrusts of each of the four 48 inch (1.2 m) diameter impellers which, due to differential pressures, create a thrust toward the suction.

This compressor exhibits the same thermal expansion and thrust forces as a steam turbine. Thrust probes are within 8 inches (203 mm) of the thrust bearings. They look at the end of the rotor and are gap increasing as active thrust increases (more negative dc volts).

Case Histories

Keeping the above in mind, the following case histories show how data received from properly mounted thrust probes alerted plant personnel to different machine problems.

Case History #1 — Salt contamination in a steam turbine in an ethylene plant

An 18,000 horsepower steam turbine in an ethylene plant was driving two centrifugal compressors at approximately 4,500 rpm. Dual thrust probes were mounted on the turbine but without automatic thrust travel shutdown limits. The thrust and radial bearing failure (Figure 3) was part of a sequence which started in the mid 1970s. As a result of this failure, the thrust axial movements were placed on automatic shutdown. The babbitt (white metal) thickness was 30 mils (762 μm). Alarm setpoints were set at **15 mils (381 μm) for Alert and 25 mils (635 μm) for Danger with automatic shutdown and a three-second integrated time delay.**

The problem

The steam conditions in this plant had been excellent for years and used flash condensers and properly treated boiler feedwater. However, two water exchangers (brine and treated water) existed within the flash evaporators.

These exchangers leaked, contaminating the steam quality with sodium chloride. The ten stage turbine developed salting problems when brine and treated water contaminated the boiler feedwater due to leaks in the internal exchangers of a flash evaporator. The salt, primarily sodium chloride (table salt), settled out selectively on the fifth and sixth stage blading, obstructing the flow to a 40 percent, or less, flow passage area. The dew point was 500-525°F (260-274°C).

The back pressure steam turbines in this plant slowed down when there was salting present. In many cases, back pressure steam turbines can reach dew point on salt and silica after exiting the turbine. This steam turbine, with extraction/condensing capabilities, had reserve steam power. What I mean is, the inlet valves continued to put more steam into the steam turbine, to hold speed, even though fouling in the fifth and sixth blading rows was increasing the load on the bearing system.

As stated above, increased load is normally indicated by:

- Increased first stage pressure (pressure behind the first nozzles). Pressure in the velocity compounded Curtis Stage. For example, steam is restricted from passing through the blading due to salt build-up.
- Increased bearing metal temperatures on the "active" thrust bearing during failure. This means the rotor is pushing in the direction of the turbine exhaust.
- Increased active thrust recordings on axial displacement monitors.

Thrust overload was present. More steam could be pushed through the turbine than the turbine was rated to handle. The thrust shutdown alarm sounded. When the vibration alarms sounded twenty seconds later, as the turbine blading contacted the diaphragms, the operator shut down the turbine. ►

This failure melted the thrust bearings. The rotor thrust face had "severe" heat check tears in the active face. The radial bearing overheated and melted the babbitt [high tin babbitt melts at 415°F (213°C)]. The melted babbitt flowed down the drain oil pipe and solidified, blocking the oil flow. It is fortunate the oil flow was blocked, otherwise plant operators would have restarted the steam turbine if oil flow could have been established.

Solution

The rotor was cleaned. The contamination to the boiler feedwater was corrected through exchanger repairs. Steam conductivity meters and alarms were installed to monitor steam conditions.

The blades were straightened in the turbine builder's repair shop. The turbine was returned to service within one week, but the failure cost over one million dollars in 1977 dollars. Normally, this type of failure is not repairable and all internal elements must be replaced.

Until this incident, this ethylene plant had refused to put the unspared main drive trains on automatic shutdown. Within a month of this machine failure, the entire plant had all unspared drive trains on automatic shutdown.

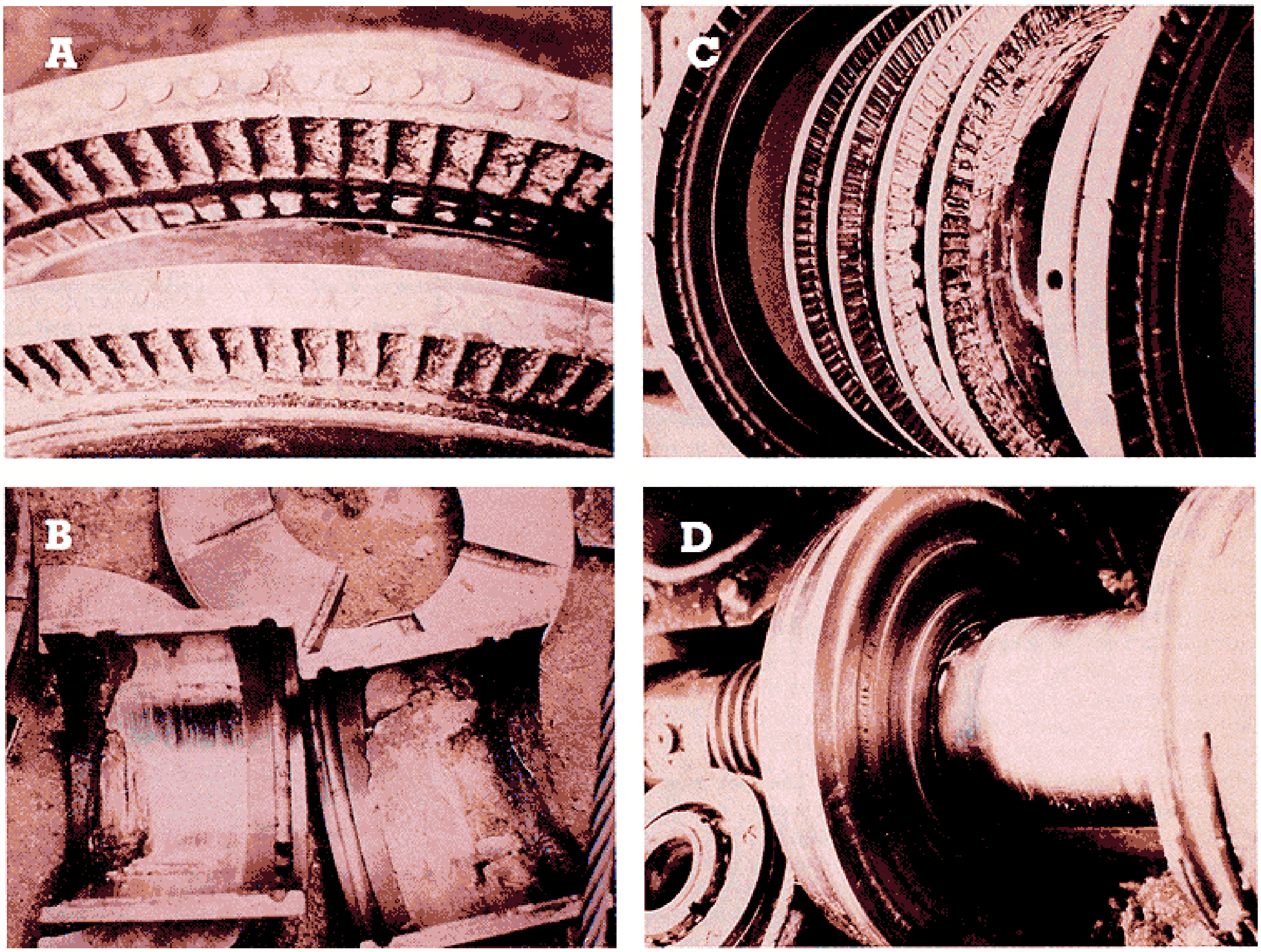

Figure 3

(A) Closeup of the fifth and sixth stage blading showing salt buildup which obstructed flow to 40 percent.

(B) Melted journal bearings at the thrust end. The melted babbitt flowed down the drain oil pipe and resolidified.

(C) Salt formation on the fifth and sixth rows of a ten stage 18,000 HP steam turbine.

(D) Damaged active thrust collar. Note: heat tears in the metal and blue color on the journal.

Case History #2

This case history involves three identical machine trains, consisting of a 4300 rpm 10,000 horsepower steam turbine driver and a centrifugal air compressor.

At 25 mils (636 μm), these machines will shut down automatically. The "cold float" on the turbines is 16 mils (406 μm). The cold float on the compressors is 17 mils (432 μm), i.e., active to inactive "bump" check. The cold float zone (Figure 4) is measured when the machine is at rest and cold (ambient temperature). The unit had **all active thrust bump/hold positions set on +7 mils (178 μm) "normal" positions.** The inactive bump can be 9 mils (229 μm) counter, 10 mils (254μm) counter, etc. The thrust range is 80 mils (2 mm) plus the linear range by specification and practice. At this plant, no attempt is made to put the zero of the monitor on "**center of float**." It is easier to hold the rotor against the active thrust bearing, than to hope it stays in the "center" of float. All the monitors are set for normal operation at +7 mils (178 μm) so a deviation is easy to detect. The voltage of the probe gap must agree with the +7 mils (178 μm) position and may well be –9 Vdc or –9.4 Vdc. The instrument builder does not give infinite latitude of adjustment. **Once the monitor has been set up initially, it is not necessary to change it during future maintenance.** Simply reset the probe gap/voltage to agree with the monitor, e.g. +7 mils (178 μm) in this case. ►

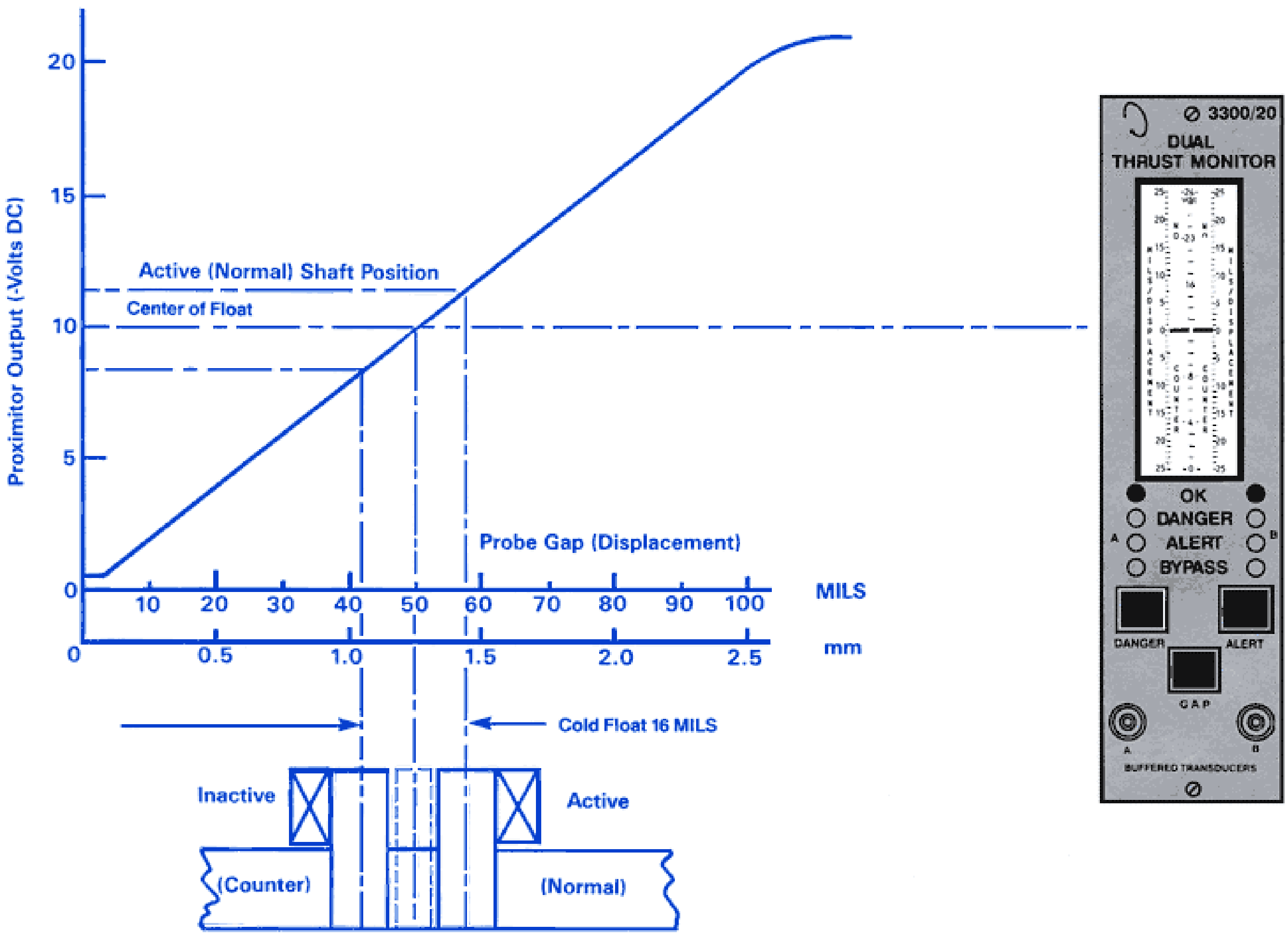

Figure 4
Center zero equals thrust collar in center of cold float zone as is shown on 3300/20 front panel display.

Each of these compressors supplies a critical reactor. No failures have occurred in fourteen years and they have not been opened. They have no fouling, with triple filters at 0.6 inch (15 mm) water pressure drop. Vibration is still in the .5 to 1.0 mils (12.7 to 25.4 μm) peak-to-peak range and the thrust on the compressors is still functioning properly.

The problem

Several incidents have occurred on the steam turbine. However, the thrust automatic shutdown has saved the equipment. One of these incidents resulted from the failure of the desuperheating station upstream of these turbines as well as other process equipment. The turbines were shut down when thrust movement reached 25 mils (635 μm). Some steam gaskets were even blown out from the drop in superheat and resulting "steam/water hammer."

A 1989 incident is a good example of poor conditions leading to back pressure turbine salting and overloading of the thrust bearing on the steam turbine. A turbine received desuperheated steam at 650° to 750° (343° to 399°C), not the traditional 600 psi (4,136,880 N/m^2) steam at 700°F (371°C). Usually, the turbine received steam at too low a superheat, between 600° and 650°F (316° to 343°C), so the maximum amount of condensation is returned to the unit's reactor steam generating system. Mineralized condensate from the deaerator and other units also added to the problem.

Eight years previously, exhaust pressure on the steam turbine had been lowered from 55 psi (379,214 N/m^2) to 30 psi (206,844 N/m^2) with the manufacturer's approval. This led to further increased thrust movement of 4 to 5 mils (102 to 127 μm) above the alarm limits and nearly to shutdown limits. Prior to this incident, first stage pressure, relative to total throttle flow and horsepower, were not plotted.

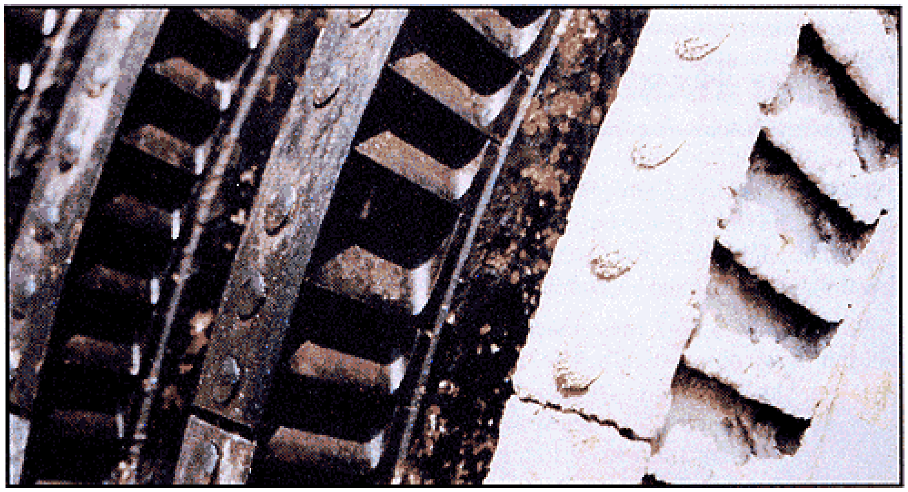

Figure 5
Salt deposits on last row of Rateau blading.

Nevertheless, the normal first stage pressures could be measured and compared to the manufacturer's charts. The reactors were scheduled to be shut down for catalyst change before the thrust shutdown limit (25 mils or 635 μm) was reached.

Steam Turbine Unit 2 was opened to install an improved radial bearing and to repair casing leaks. The last row of 5 Rateau blading was heavily fouled (Figure 5). The thrust bearing shows the overloaded "darkened" areas, confirming that 75% arc and 75% radius remains an accurate placement for the thermocouple. (However, never embed thermocouples into the babbitt!)

Solution

The rotor was inspected through the tapped hole for the Sentinel valve, which fortunately looks directly at the last stage bucket (on center pitch). Blade conditions were recorded with a lens/light through the plug for the Sentinel valve. Two selected hydroblast nozzles with reduced water pressure were used to clean the buckets through this opening without disassembling the steam turbine. The area was washed to very good condition while spinning the 10,000 horsepower steam turbine rotor at 70 rpm.

Conclusions

When setting shutdown limits, never try to save the thrust bearing. **Save the machine. Setting short limits only causes several shutdowns where *no damage is seen in the thrust bearings,*** resulting in nuisance shutdowns. ***These nuisance shutdowns result in the automatic shutdown feature being defeated.*** Automatic thrust shutdown has prevented many $1 to $2 million losses.

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